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DESIGN AND FINITE ELEMENT ANALYSIS OF COMPOSITE MATERIAL PRESSURE VESSELS

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ABSTRACT

Pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. Information such as design and component development time was analyzed and modeled to ensure the effect of implementation of this approach to product development cycle and design efficiencies This project discusses some design principles that are deals with vessels are subjected to various applied forces acting in combination with internal pressure. Design of pressure vessels is governed by the ASME pressure vessel code. The code gives for thickness and stress of basic components, it is up to the designer to select appropriate analytical as procedure for determining stress due to other loadings. Structures such as pipes or bottles capable of holding internal pressure have been very important in the history of science and technology. Design of different pressure vessel concerned with elements such as shell, Dish end, and nozzles based on standards and codes; and evolution of shell, dish end and nozzles analyzed by means of ANSYS .for three materials (composite material, aluminum alloys and stainless steel) and then compere to choose the best design

KEYWORDS: Pressure Vessels Design ASMA, Modeling CATIA – V5 Finite Element Analysis ANSYS

1- INTRODUCTION

A pressure vessel is a closed container designed to hold gases or liquids at a pressure substantially different from the ambient pressure. The pressure differential is dangerous and fatal accidents have occurred in the history of pressure vessel development and operation. Consequently, pressure vessel design, manufacture, and operation reregulated by engineering authorities backed by legislation. For these reasons, the definition of a pressure vessel varies from country to country, but involves parameter such as maximum safe operating pressure and temperature. The pressure vessels are used to store fluids under pressure. The fluid being stored may undergo a change of state inside the pressure vessels as in case of steam boilers or it may combine with other reagents as in chemical plants. Pressure vessels find wide applications in thermal and nuclear power plants, process and chemical industries, in space and ocean depths, and in water, steam, gas and air supply system in industries, From an engineering point of view, properties connected with metals are elasticity, plasticity, brittleness, malleability and ductility . Many of these properties are contrasting in their nature so that a given metal cannot exhibit simultaneously all these properties, So the design engineers should dealing carefully at the selection of the materials that used for the design .In our project we have taken three materials (ferrous , non-ferrous and **composite materials**) and then explaining the difference among them. Composite materials have become common engineering

materials and are designed and manufactured for various applications such as pressure vessels.

2- TYPES OF PRESSURE VESSELS

Following are the main types of pressure vessels According to the end construction According to the dimensions Pressure vessel according to the end construction According to the end construction, the pressure vessels are may be open of end or closed end .A simple cylinder with a piston is an example of open–end vessel whereas a tank is an example of closed end vessel .Due to fluid pressure circumferential stresses are induced in case of closed ended vessels.

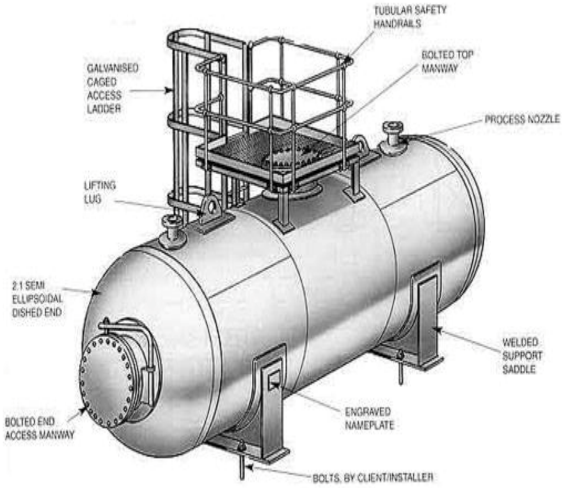


Figure 1

According to the dimensions pressure vessels may be of thin or thick shell .The deciding factor among thin and thick shells is its wall thickness and shell diameter if the ratio t/D is less than 1/10 the vessel is said to be thin shell and if the ratio is greater than 1/10 it is said to be thick shell. Thin shells are used in boilers, tanks and pipes whereas thick shells are used in high pressure cylinder.

3-GENERAL THEORETICAL DESIGN

There are some variations of the basic equations in various design codes. Some of the equations are based on thick-wall Lame equations .In this work such equations will be discussed:

- **Thin-Shell Equations**

Let us consider a long thin cylindrical shell of radius R and thickness t, subject to an internal pressure p. By thin shells we mean the ones having the ratio R/t typically greater than about 10. If the ends of the cylindrical shell are closed, there will be stresses in the hoop as well as the axial (longitudinal) directions. A section of such a shell is shown in Figure 3.2 The hoop (circumferential) stress, $\sigma\ hoop$ and the longitudinal stress,$\sigma\ long$ are indicated in the figure.

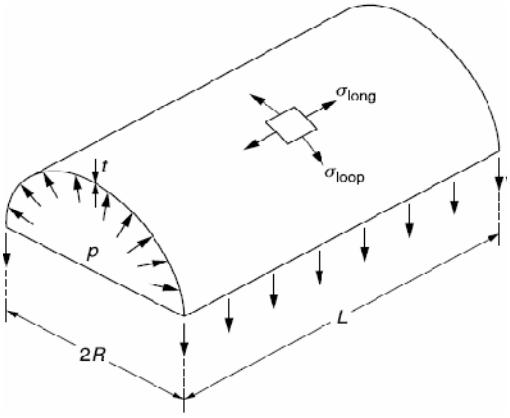


Figure 2

$\sigma\,\text{hoop} = \sigma\theta = \frac{N\theta}{t} = \frac{P\,R}{t}$ ……..1

And the longitudinal stress:

$\sigma\,\text{long} = \sigma\,\varphi = \frac{N\varphi}{t} == \frac{P\,R}{2t}$ ..… .2

- **Thick-Shell Equations**

For R/t ratios typically Less than 10, Eqs. (1) and (2) tend not to be accurate, and thick-shell equations have to be used radial and hoop stresses

$$\sigma_{\text{rad}} = \frac{P}{(m^2-1)}\left[1 - \frac{R_o^2}{r^2}\right]$$

$$\sigma_{\text{hood}} = \frac{P}{(m^2-1)}\left[1 + \frac{R_o^2}{r^2}\right]$$

$$\sigma_{\text{long}} = \frac{PR_i^2}{(R_o^2-R_i^2)} = \frac{P}{(m^2-1)}$$

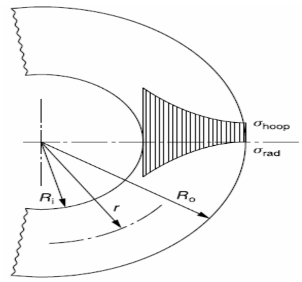


Figure 3

4-STUDY CASE

In this work we take special case, where we have taken horizontal pressure vessel under internal pressure by using ASME Section VIII Division I .I and we explained the main parts which consist of Shell hemispherical head nozzle (inlet, outlet, manhole and drain)

Design Data

Horizontal Pressure vessel

Length of Pressure vessel, L = 5.5 m

Inside Diameter, Di = 1.8 m

Internal Pressure = 2 MPa

E: longitudinal joint efficiency =1

E: circumferential joint efficiency = 0.85

Case No. 1 the material Stainless steel (SA- 516 G 70)

S: allowable stresses = 120.66 MPa

C.A: corrosion allowance = 1.588 mm/y

Manufacturing tolerance of material = 0.125

E_1: Modulus of elasticity = 202.707 Gpa

υ : Poisson ratio = 0.3

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints)

$\mathrm{t} = \frac{\mathrm{P\ Ri}}{\mathrm{S\ E - 0.6\ P}} + \mathrm{C.A} = \mathrm{t} = 16.7\ \mathrm{mm}$

Longitudinal Stress (circumferential joints)

$\mathrm{t} = \frac{\mathrm{P\ Ri}}{\mathrm{2\ S\ E + 0.4\ P}} + \mathrm{C.A} = \mathrm{t} = 10.33\ \mathrm{mm}$

The minimum required thickness (Governing). Adopted shell thickness, t = 17 mm. where $\mathrm{t} < 0.5\,Ri$ and maximum allowable working pressure, p not exceed 0.385 SE. There for the design thickness in safety

$\sigma\mathrm{hoop} = 106.88\ \mathrm{MPa}$

$\sigma\ \mathrm{long} = 52.5\ \mathrm{MPa}$

Hemispherical Head Design of Pressure Vessel

$\mathrm{t} = \frac{\mathrm{P\ Ri}}{\mathrm{2SE - 0.2P}} + \mathrm{C.A} = \mathrm{t} = 9.06\ \mathrm{mm}$

The minimum required thickness (Governing). Adopted shell thickness, t = 9.1 mm .where the ASME estimate is conservative in this case.

$\sigma\ \mathrm{hoop} = \sigma\ \mathrm{long} = 98.908\ \mathrm{MPa}$

Case No. 2 the Aluminum alloys material (6061-T6)

S: allowable stresses = 75 .152 MPa

C.A: corrosion allowance = 0.65 mm/year

Manufacturing tolerance of Aluminum alloys (6061 – T6) = 0.125

E_1: Modulus of elasticity = 70 GPa

υ : Poisson ratio = 0.33

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints)

$t = 25 \text{ mm}$

Longitudinal Stress (circumferential joints

$t = 14.65 \text{ mm}$

The minimum required thickness (Governing). Adopted shell thickness, t = 25 mm. where $t < 0.5\, Ri$ and maximum allowable working pressure, p not exceed 0.385 SE. There for the design thickness in safety

$\sigma\text{hoop} = 73 \text{ Mpa}$

$\sigma \text{ long} = 35.51 \text{ MPa}$

Hemispherical Head Design of Pressure Vessel

$t = 12.66 \text{ mm}$

The minimum required thickness (Governing). Adopted shell thickness, t = 12.66 mm .where the ASME estimate is conservative in this case.

$\sigma \text{ hoop} = \sigma \text{ long} = 71.1 \text{ Mpa}$

Case No. 3 the Aluminum / Silicon Carbide material (MMCs)

S: allowable stresses = 163.7 MPa

C.A: corrosion allowance = 1.62 mm/year

Manufacturing tolerance of Aluminum / Silicon Carbide (MMCs) = 0.125

E_1: Modulus of elasticity = 81 GPa

υ : Poisson ratio = 0.3

Design Shell of Pressure Vessel

Circumferential stress (Longitudinal Joints

$t = 12.7 \text{ mm}$

Longitudinal Stress (circumferential joints

$\mathrm{t} = 8.1\ \mathrm{mm}$

The minimum required thickness (Governing). Adopted shell thickness, t = 12.7 mm. where $\mathrm{t} < 0.5\,Ri$ and maximum allowable working pressure, p not exceed 0.385 SE. There for the design thickness in safety.

$\sigma\mathrm{hoop} = 142.73\ \mathrm{Mpa}$

$\sigma\ \mathrm{long} = 70.37\ \mathrm{MPa}$

Hemispherical headdesign of pressure vessel

$t = 7.12\ mm$

The minimum required thickness (Governing). Adopted shell thickness, t = 7.12 mm .where the ASME estimate is conservative in this case.

$\sigma\ \mathrm{hoop} = \sigma\ \mathrm{long} = 126.41\ \mathrm{MPa}$

Case No. 1 (Stainless steel SA 516 G 70)

Table 1

No.	PARTS		THICKNESS, mm	LENGTH, mm	DIAMETER, mm	
					IN	OUT
1	Shell		17	5500	1800	1834
2	Hemispherical Head		9.1	900	1800	1818.2
3	Nozzle(inlet and outlet)		20.6	203.2	177.88	219.08
4	Reinforcement for Inlet and outlet nozzle	**A1**	4.3	--------	177.88	355.76
		A21	19.1	48.5	180.88	219.08
		A22	20.6	48.5	177.88	219.08
5	Nozzle manhole		20.6	254	466.8	508
6	Reinforce for man hole	**A1**	4.3	----------	466.8	933.6
		A21	16.69	48.5	474.62	508
		A22	20.6	48.5	466.8	508
7	Drain nozzle		22.23	152.4	82.54	127
8	Reinforce for Drian	**A1**	4.3	-------	82.54	156.8
		A12	21.53	48.5	83.94	127
		A22	22.23	48.5	82.54	127

Case No. 2 (ALUMIMUM ALLOYS 6061- T6)

Table 2

NO	PARTS		THICKNESS, mm	LENGTH, mm	DIAMETER, mm	
					IN	OUT
1	Shell		25	5500	1800	1850
2	Hemispherical Head		12.66	900	1800	1825.32
3	Nozzle(inlet and outlet)		28.6	203.2	161.88	219.08
5	Reinforcement for Inlet and outlet nozzle	**A1**	4.26	---------	161.88	323.76
		A21	26.41	71.5	166.26	219.08
		A22	28.6	71.5	161.88	219.08
6	Nozzle man hole		32.54	254	442.92	508
7	Reinforce for man hole	**A1**	4.26	---------	442.92	885.84
		A21	26.54	71.5	454.92	508
		A22	32.54	71.5	442.92	508
8	Drain nozzle		28.6	152.4	69.8	127
9	Reinforce for Drian	**A1**	4.26	--------	69.8	184.2
		A12	27.656	71.5	71.688	127
		A22	28.6	71.5	69.8	127

Case No.3 (COMPOSITE MATERIAL ALSIC)

Table 3

NO	PARTS		THICKNESS, mm	LENGTH, mm	DIAMETER, mm	
					IN	OUT
1	Shell		12.7	5500	1800	1825.4
2	Hemispherical Head		7.12	900	1800	1814.24
3	Nozzle(inlet and outlet)		15.06	203.2	188.96	219.08
5	Reinforcement for Inlet and outlet nozzle	**A1**	3.43	-------	188.96	219.08
		A21	13.897	36.275	191.28	219.08
		A22	15.06	36.275	188.96	219.08
6	Nozzle man hole		15.06	254	477.88	508
7	Reinforce for man hole	**A1**	3.43	----------	477.88	955.76
		A21	12.12	36.275	483.76	508
		A22	15.06	36.275	477.88	508
8	Drain nozzle		15.9	152.4	95.2	127
9	Reinforce for Drian	**A1**	3.43	-------	95.2	190.4
		A12	15.31	36.275	96.38	127
		A22	15.9	36.275	95.2	127

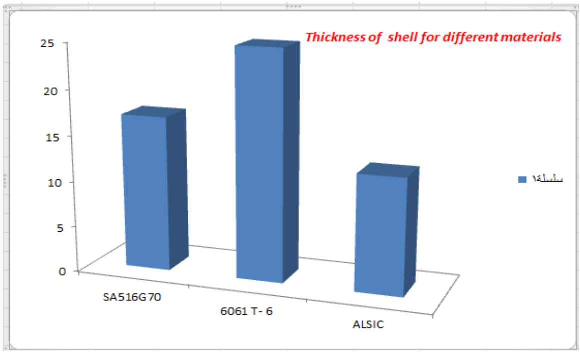


Figure 4

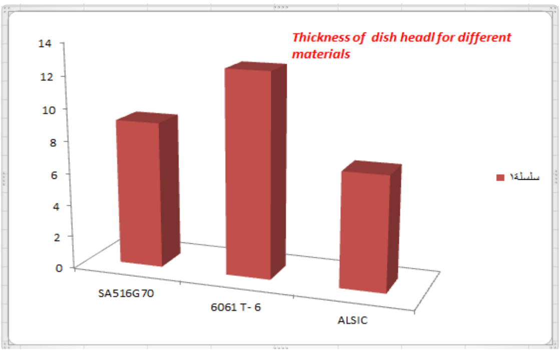


Figure 5

5- MODELING

Modeling provides the design engineer with intuitive and comfortable modeling techniques such as sketching,

Stainless Steel SA 516 G 70

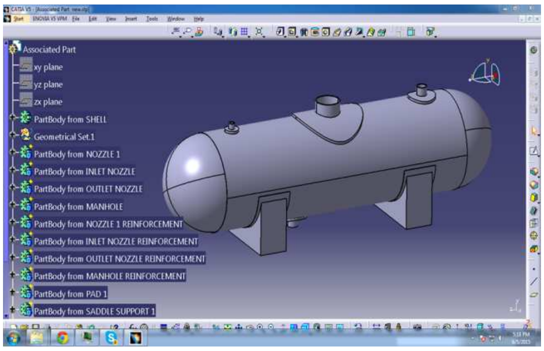


Figure 6

ALUMIMUM ALLOYS 6061- T6

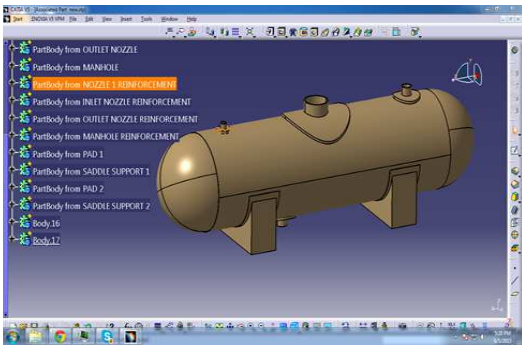


Figure 7

COMPOSITE MATERIAL ALSIC

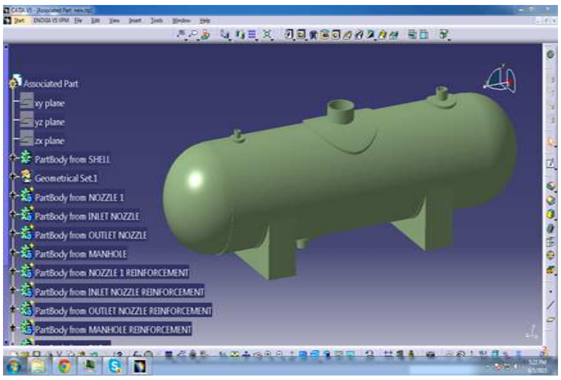


Figure 8

6-FINITE ELEMENT ANALYSIS

In this study, finite element stress was performed to investigate the longitudinal stresses, hoop stresses and nozzle stresses of an pressure vessel set as mentioned above. The geometrical model of a pressure vessel was generated using parametric equation. Commercial FEA software, ANSYS workbench 14.5.7,.

The longitudinal stresses, for three material of a pressure vessel calculated by FEA ANSYS is close longitudinal stresses obtained from calculation using (ASME VIII) stress formula and the longitudinal stresses are below the allowable limited. Hence the pressure vessel is safe in longitudinal stress. In these tables shown below the longitudinal stresses

obtained from the analysis of the three- dimension model using ANSYS WORKBANCH 14.5.7 compared with the calculated values from the standard by (ASME VIII).

Table 4

shell	Internal Pressure MPa	longitudinal stress MPa ANSYS	Longitudinal stress MPa ASME	Difference ratio
SA 516 G 70	2	56.501	52.5	7.1%
Aluminium alloys 6061 T- 6	2	37.204	35.51	4.5%
ALSI C	2	68.666	70.37	2.4%

Compared ASME value with ANSYS value of shell

Table 5

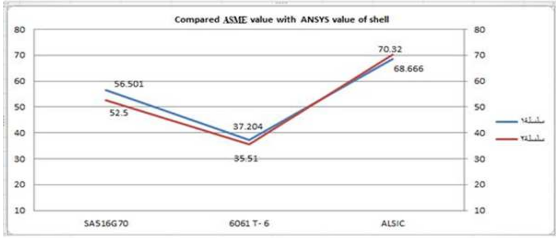
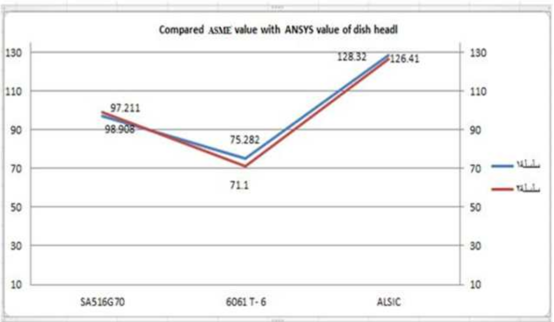


Table 6

Dish head	Internal Pressure MPa	longitudinal stress MPa ANSYS	longitudinal stress MPa ASME	Difference Ratio
SA 516 G 70	2	97.211	98.908	1.7%
6061 T- 6	2	75.282	71.1	5.9%
ALSI C	2	128.320	126.41	1.5%

Compared ASME value with ANSYS value of dish head

Table 7



The hoop stresses, for three material of a pressure vessel calculated by FEA ANSYS is close hoop stresses obtained from calculation using (ASME VIII) stress formula and the hoop stresses are below the allowable stress. Hence the pressure vessel is safe in long stress. The tables (7.3 -7.4) the hoop stresses obtained from the analysis of the three- dimension model using ANSYS WORKBANCH 14.5.7 compared with the calculated values from the standard recommended by (ASME VIII) during the design of pressure vessel

Table 8

shell	Internal Pressure MPa	hoop stress MPa ANSYS	hoop stress MPa ASME	Difference Ratio
SA 516 G 70	2	103.769	106.88	2.9%
6061 T 6	2	75.624	73	3.5%
AL SI C	2	144.950	142.73	1.5%

Compared ASME value with ANSYS value of dish head

Table 9

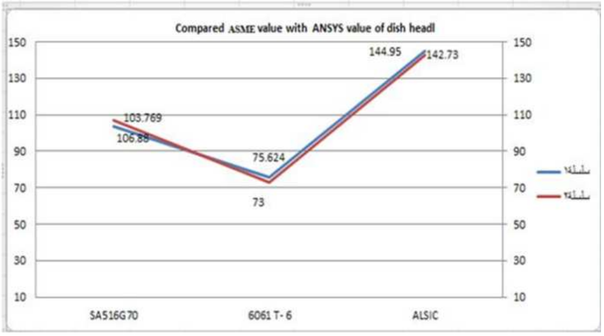
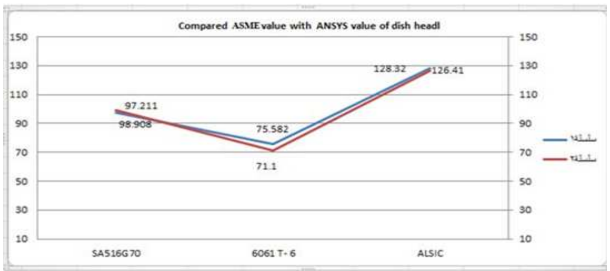


Table 10

Dish head	Pressure MPa	hoop stress MPa ANSYS	hoop stress MPa ASME	Difference Ratio
SA 516 G 70	2	97.211	98.908	1.7%
AL 6061 T 6	2	75.582	71.1	5.9%
AL SI C	2	128.320	126.41	1.5%

Compared ASME value with ANSYS value of dish head

Table 11



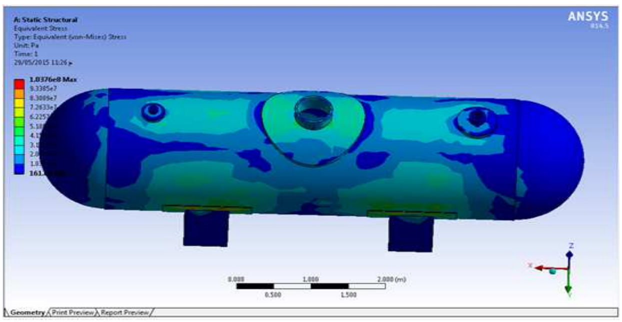


Figure 9

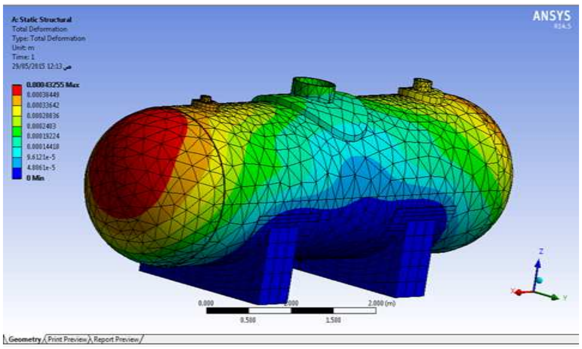
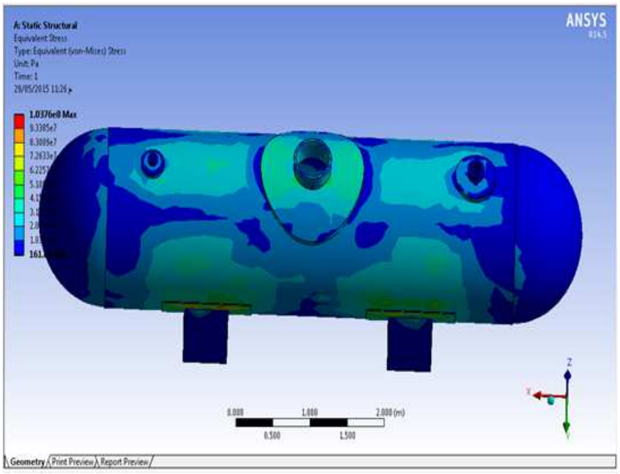


Figure 10



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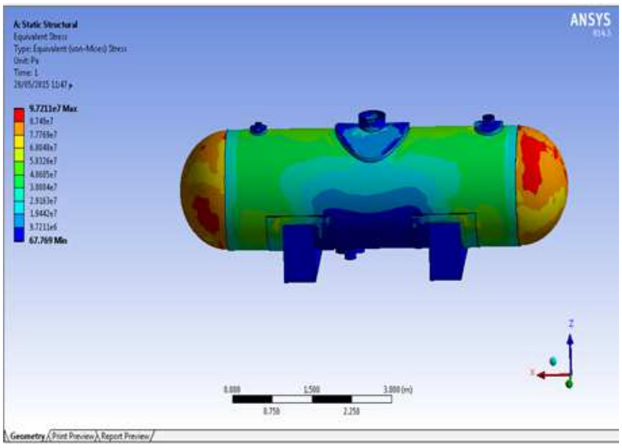

Figure 12

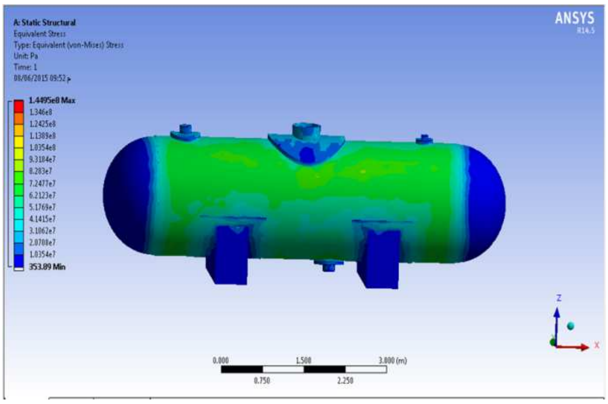

Figure 12

7- CONCLUSIONS

The empirical approach corresponds to design by rule and finite element analysis corresponds to design by analysis method are adopted and calculations were made according to (ASME pressure vessel code).

The stress distribution of various geometric parameters of each part is observed to select the optimal thickness of three materials. This shows that the design by analysis is the most desirable method to evaluate and predict the behavior of different configurations of pressure vessel.

The comparison of' these results helps to provide the most optimized design with an ability to meet the requirements. After analyzing the stress behavior of the pressure vessel with different geometrical parameters, it is concluded that the design of given Horizontal pressure vessel is safe according to the both the results.

The stress values obtain by empirical method and analysis stresses are below allowable limits which are acceptable. When we got the results have chosen best design among three materials where we found the composite material thickness was 12.7 mm of shell and 7.12 mm of hemispherical head also the longitudinal and hoop stress are below allowable limits. Therefore the gusset is achieved. Which leads to saving material and weight with it.

The longitudinal stresses, for three material of a pressure vessel calculated by FEA ANSYS is close longitudinal stresses obtained from calculation using (ASME VIII) stress formula and the longitudinal stresses are below the allowable limited. Hence the pressure vessel is safe in longitudinal stress,

The hoop stresses, for three material of a pressure vessel calculated by FEA ANSYS is close hoop stresses obtained from calculation using (ASME VIII) stress formula and the hoop stresses are below the allowable stress. Hence the pressure vessel is safe in long stress,

The MAX contact stresses, for three material of an nozzle (inlet and out let , manhole ,drain) pressure vessel calculated by FEA ANSYS is close max contact stresses obtained from calculation using (ASME VIII stress formula and The stresses are below the allowable stress. Hence the pressure vessel is safe in nozzle stress.

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